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# A VACUUM FOUR-BALL TRIBOMETER TO EVALUATE LIQUID LUBRICANTS FOR SPACE APPLICATIONS

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## SUMMARY

The design and operation of a vacuum tribometer, based on the four-ball configuration, is described. This tribometer evaluates the tribological characteristics of liquid lubricants for space applications. Operating conditions include: room temperature, loads to  $\sim 1000\text{N}$ , speeds to  $\sim 500$  rpm, and pressures of  $\sim 10^{-6}$  Pa. Tests can also be run at atmospheric pressure with air or nitrogen. Some typical test results are included.

## INTRODUCTION

The advent of new satellite, spacecraft, and space station components will place increased burdens on the lubrication systems for the many mechanical moving assemblies (ref. 1). These assemblies include: momentum/reaction wheels, solar array drives, pointing mechanisms, de-spin mechanisms, slip rings, gears, etc. (ref. 2). Improved lubrication systems are not only required because of increased mission lifetimes but also to insure greater reliability. In the past, other systems (e.g., batteries, electronics, thermal and optical systems) caused premature spacecraft failures (ref. 3). It is now apparent, that advances in these areas have exposed tribology as the primary roadblock in achieving mission requirements.

Liquid lubricants (or greases) are often used in space mechanisms for a variety of reasons. These include: no wear in the elastohydrodynamic range, low mechanical noise, ease of replenishment, relatively insensitive to environment, and ability to scavenge wear debris. However, there have been few tribological studies (ref. 4 to 6) of their behavior in air and vacuum at high loads.

There are a number of tribometers (e.g., pin-on-disk, Falex, Timken, SAE, etc.) used to evaluate liquid lubricants under high contact pressure conditions (ref. 7). Another device, known as a 4-ball apparatus, is commonly used (ref. 8). It utilizes a rotating ball loaded and sliding against three stationary balls immersed in the lubricant. This device has the advantages of simplicity, high load capability and readily accessible wear specimens (standard bearing balls).

Therefore, the objective of this paper is to report the design and operation of a vacuum tribometer for liquid lubricants based on the four-ball geometry. Some typical test data are included.

## VACUUM FOUR-BALL TRIBOMETER

### Overall Description

The overall apparatus is shown in figure 1. The specimen configuration of the apparatus is essentially the same as the ordinary four-ball apparatus (ref. 8) as shown in figure 2, except for the use of 9.5 mm (3/8 in.) diameter precision bearing balls (grade 10). The apparatus is mounted in a vacuum chamber so that tests can be carried out under high vacuum. The vacuum chamber is evacuated using a turbomolecular pump (140 l/s) and a mechanical backing pump to achieve a vacuum of approximately  $10^{-4}$  to  $10^{-6}$  Pa. All flanges used for the vacuum chamber are knife-edge flanges with copper gaskets except for a quick access door which uses an elastomeric O-ring. The vacuum chamber is equipped with a hot filament ionization gage for chamber pressure and a mass spectrometer (residual gas analyzer).

The rotating upper ball is mounted on a spindle which is connected to a ferrofluidic rotary feedthrough. The lower three stationary balls are fixed in a ball holder (lubricant cup) which is mounted on the stage. The stage can be moved upward from outside the chamber with a pneumatic cylinder through a linear motion feedthrough sealed with a welded metallic bellows.

The shaft of the linear motion feedthrough is supported under the "flex-pivot" inside the vacuum chamber with a linear ball bearing (ball bushing) lubricated with  $\text{MoS}_2$  powder coating. The lower end of the shaft of the linear motion feedthrough is mounted on a plate outside the chamber which is supported with four linear ball bearings. A load cell is mounted between the plate and the pneumatic cylinder to measure the applied load. The axial load was calibrated under vacuum by placing another load cell between the spindle and the lower stage.

### Frictional Torque Measurement

The "flex-pivot," shown in figure 2, which is stiff toward axial thrust but elastic for angular displacement around its center axis is used to mount the stage, where the lubricant cup is fixed, on the top of the shaft of the linear motion feedthrough. This device allows the rig to have high axial applied loads (up to 1000 N) keeping a certain amount of freedom of angular displacement with low torque. For this apparatus, a "flex-pivot" having nominal torsional spring constant of  $48.6 \text{ N-m-rad}^{-1}$  was selected. Other "flex-pivots" with lower torsional spring rates can be used depending on the experimental conditions. Torque is obtained by measuring the angular displacement of the cup holding the three balls. A set of Hall-effect position sensors and a magnet are used to measure the angular displacement. Calibration torque and output voltage of the position sensor was carried out by hanging dead weights on one end of a cable which ran over a pulley and had the other end wound around the periphery of the lubricant cup.

### Capability of the Tribometer

Capability of the tribometer which was designed in this study are as follows:

atmosphere:	air, nitrogen, or vacuum
axial load range:	50 to 1032 N

rotating speed: 10 to 500 rpm  
environmental pressure:  $10^{-6}$ Pa range  
temperature: room temperature (not controlled)

### Vibration Suppression by Eddy-Current Damping

Frictional force at the tribological contact varies as a function of time. Therefore, the use of a torsionally elastic device as a flex-pivot can allow angular vibrations to be induced in this system. When this occurs, not only do friction measurements become impossible, but also the sliding conditions are altered. Eddy current damping (refs. 9 to 11) is used to suppress these unwanted dynamics.

The damping system used here consists of a yoke attached to the lower, stationary part of the flex-pivot. Magnets at the end of the yoke provide a field of 8.4 k gauss over about 1 in.<sup>2</sup>. A plate of OFHC copper, 0.25 in. thick, attached to the upper flexible part of the flex-pivot projects part-way into the magnetic field. The horizontal motion of this copper plate is damped by the magnetic field. Eddy current damping forces are proportional to the velocity of the copper plate, so that high frequency components of motion are preferentially attenuated. Thus, the unwanted vibrations of the system are suppressed, while the displacement of the flex-pivot due to steady-state values of the friction force is retained.

The effectiveness of this damping system is shown in figure 3. In figure 3(a), an oscilloscope trace of the output of the angular displacement sensor is shown after an impulse is applied to the system without eddy current damping. Strong "ringing" persists for many tens of seconds. An impulse applied to the system with the eddy current damper is damped out within about 100 msec.

### Experimental Procedure

Cleaning of the Test Balls and Lubricant Cup.—Before experiments, test balls made of AISI 440C stainless steel are cleaned by scrubbing with fine alumina powder (nominal diameter of 0.3  $\mu$ m) under stream of water followed by rinsing with deionized water. The water remaining on the ball surface is wiped off with clean filter paper. After cleaning, the balls are kept in a dry box under a dry nitrogen atmosphere.

The lubricant cup and the rotating ball holder are cleaned in the same fashion as the balls, when they are used for the first experiment with a different lubricant. Before each experiment, the cup is ultrasonically cleaned with a suitable solvent which dissolves the test lubricant. Trifluorotrchloroethane was used for removing perfluoropolyether lubricants.

### Tribology Experiment

The lubricant cup is filled with the test lubricant after the three balls are fixed in it. Then, it is placed in a glass jar and evacuated with a mechanical vacuum pump to remove dissolved air, for about one hour at room temperature under a vacuum of about 1 Pa.

After degassing, the lubricant cup is placed on the stage inside the tribometer chamber and the chamber is evacuated with a mechanical pump. After maintaining the pressure around 10 Pa for 10 min, the turbomolecular pump is engaged. This sequence minimizes bubbling and splashing of the lubricant during chamber evacuation which would cause contamination of the chamber and the mass spectrometer head.

After reaching a pressure in the range of  $10^{-5}$  Pa or less, the friction experiment is started. The frictional torque is recorded continuously throughout the experiment. Wear is determined by measuring wear scar diameters on the three stationary balls using an optical microscope after removal of the stationary cup from the chamber. A sample stage on the optical microscope is so designed that the wear scar diameter can be measured without disassembling the balls from the cup. The experiment can then be continued using the same set of balls after wear measurement, if necessary.

Figure 4 shows a typical optical micrograph of a wear scar generated under vacuum and the following conditions.

load:	200 N
speed:	100 rpm
lubricant:	perfluoropolyether (Z-25)
sliding distance:	366 m

### EXAMPLES OF DATA

Figure 5 shows some typical data on wear scar versus axial load obtained in an air environment, a rotational speed of 100 rpm, and a test duration of 30 min. The lubricant was a perfluoropolyether (PFPE) fluid (Z-25). In this figure, calculated initial hertzian contact diameters are also shown.

Figure 6 shows the friction coefficient and the wear scar diameter as a function of sliding distance. The wear scar diameter increased with increasing sliding distance. The friction coefficient showed somewhat erratic behavior but had mean value of around 0.1, after a sliding distance of about 100 m.

Figure 7 is a plot of mass intensities as a function of time for Z-25 under a 200N load and 100 rpm. Three masses were monitored: 28 (CO), 47 (COF) and 69 (CF<sub>3</sub>). CO is a gas dissolved in the lubricant. A increase in mass 28 during a test would indicate incomplete degassing of the fluid prior to the test. During the initial part of the test (up to ~1200 sec) the intensity of mass 28 is higher than background. However, as the fluid is agitated and heated by the rotating ball, the intensity approaches the background level.

In contrast, masses 47 and 69, represent lubricant breakdown products (ref. 5) generated in the tribological contact. Their intensity increases as the test proceeds. This is caused by conversion of the steel surfaces to FeF<sub>3</sub> which catalyzed further decomposition of the lubricant with attendant release of volatile species. At test conclusion, all intensities return to the levels prior to initiation of sliding.

Figure 8 contains a comparison of wear rates for three commercially available aerospace lubricants in air and vacuum. Test conditions were: 25 °C, 200N load, and a 100 rpm rotational speed. The three lubricants were (1) an unbranched perfluoropolyether (PFPE) (Z-25), (2) a branched PFPE 143AB and (3) a formulated synthetic hydrocarbon containing an antiwear compound and an antioxidant.

Results in air and vacuum clearly discriminate between the more reactive unbranched PFPE (Z-25) compared to the less reactive branched fluid (143AB). This trend correlated with other vacuum four-ball results (ref. 6) and vacuum sliding experiments (ref. 5). In addition, the better performance of formulated hydrocarbons compared to unformulated PFPE fluids correlates with oscillating gimbals tests (ref. 12) and boundary lubricant screening tests (ref. 13).

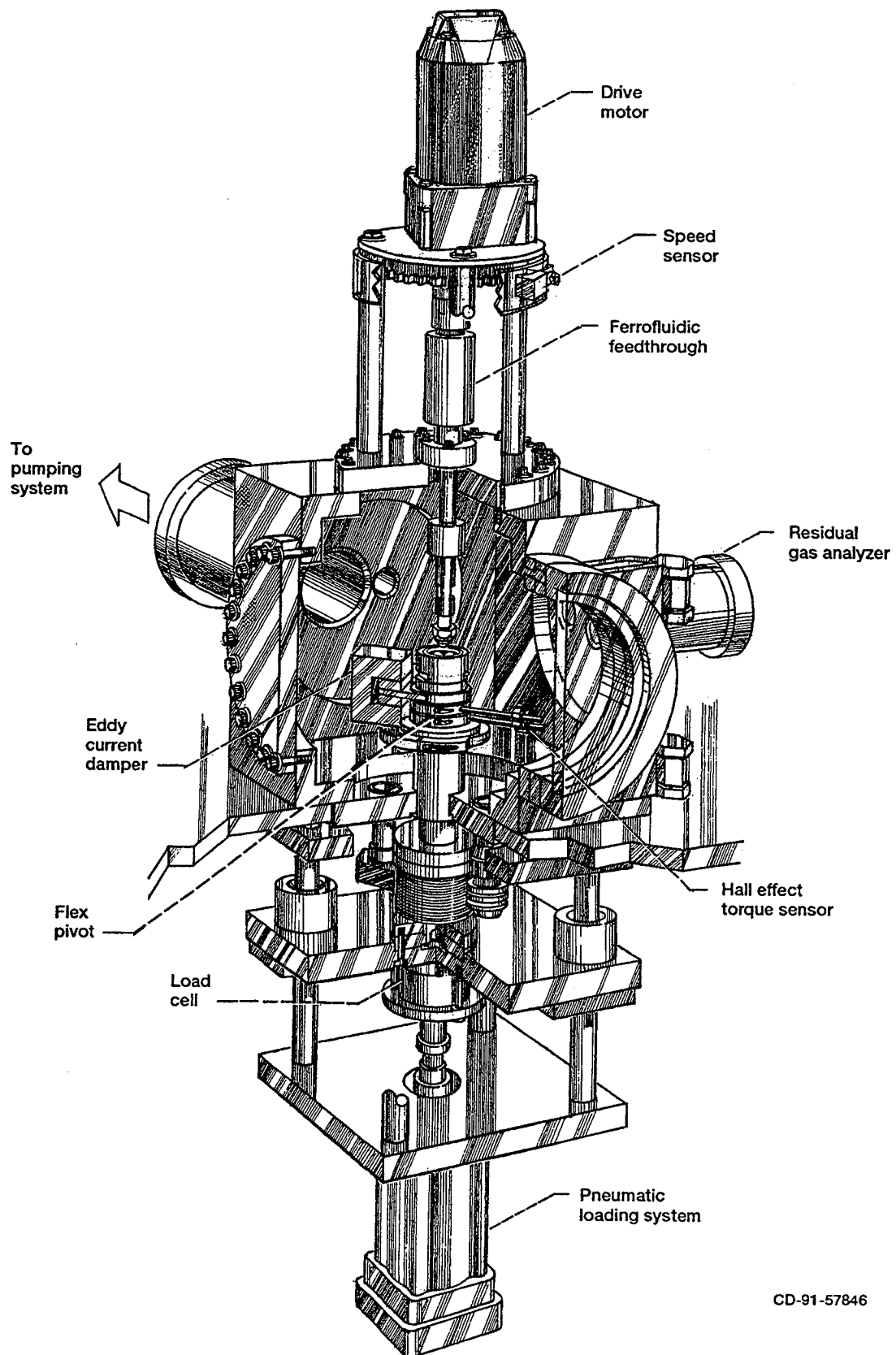


## CONCLUDING REMARKS

A newly designed vacuum four-ball tribometer for evaluating the tribological characteristics of liquid lubricants under sliding conditions at high load in high vacuum has been described. Some initial friction and wear results are presented.

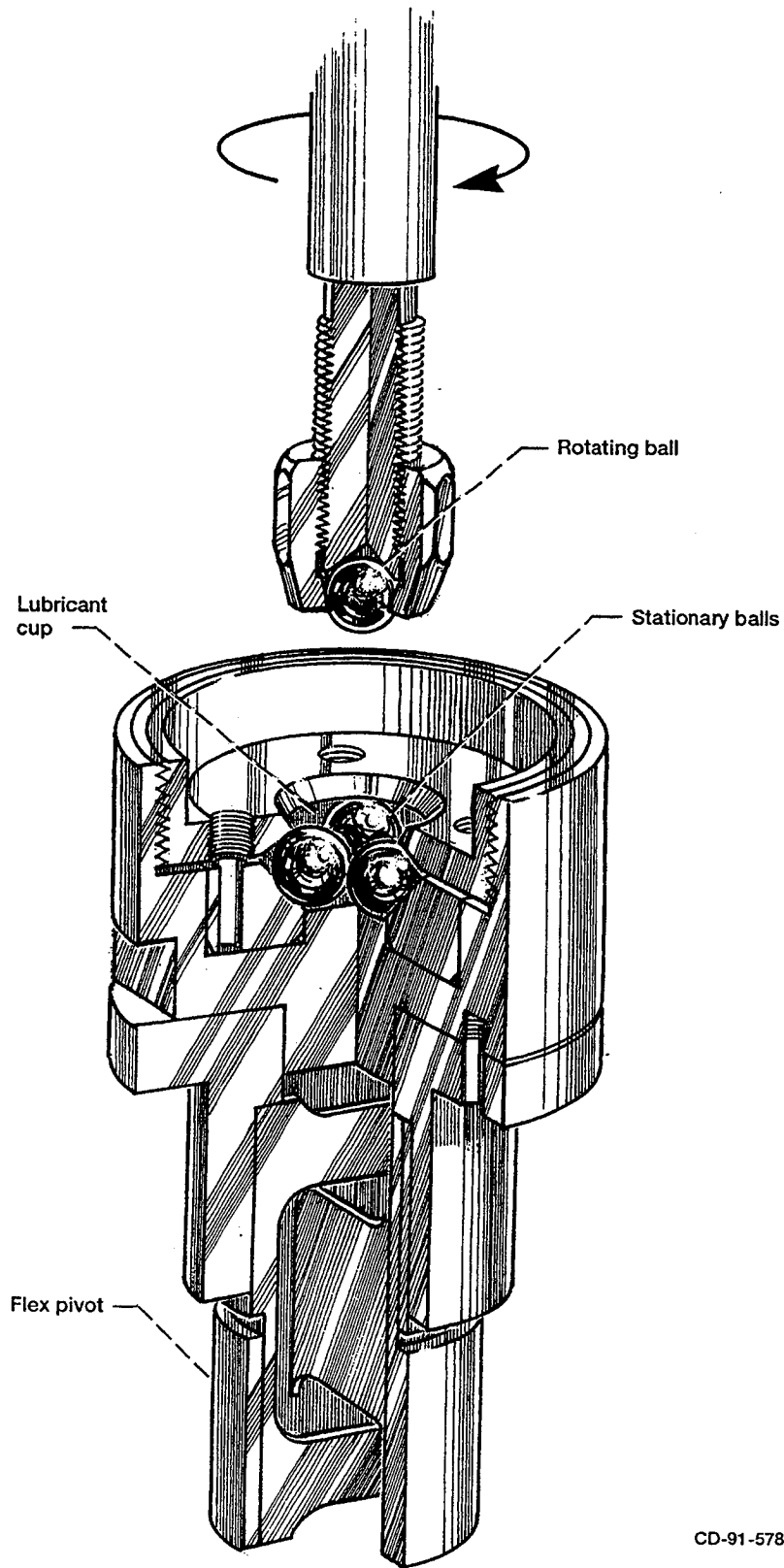
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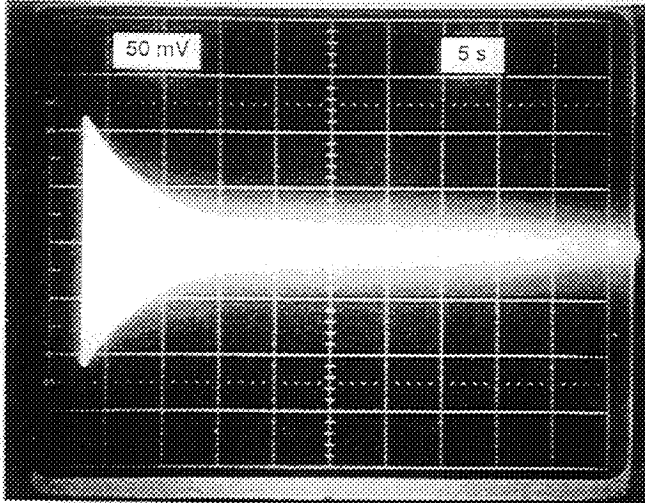
Figure 1.—Vacuum tribometer.



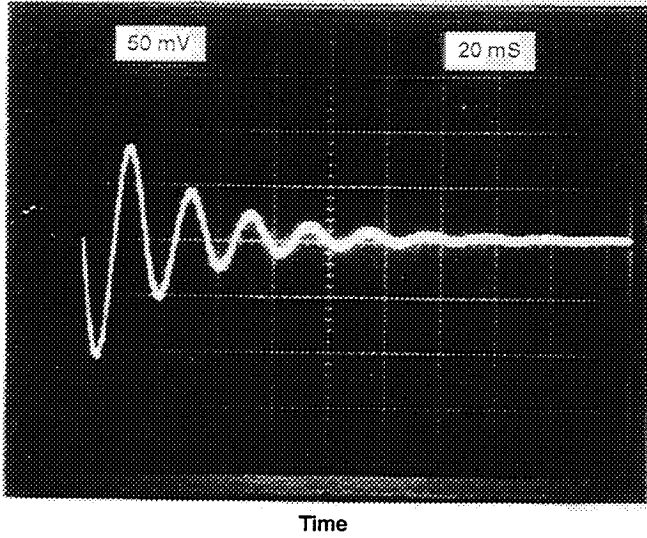
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Figure 2.—Specimen configuration.

Displacement sensor output  
(arbitrary units)



(a) Without damper.



(b) With damper.

Figure 3.—Effect of eddy-current damper.

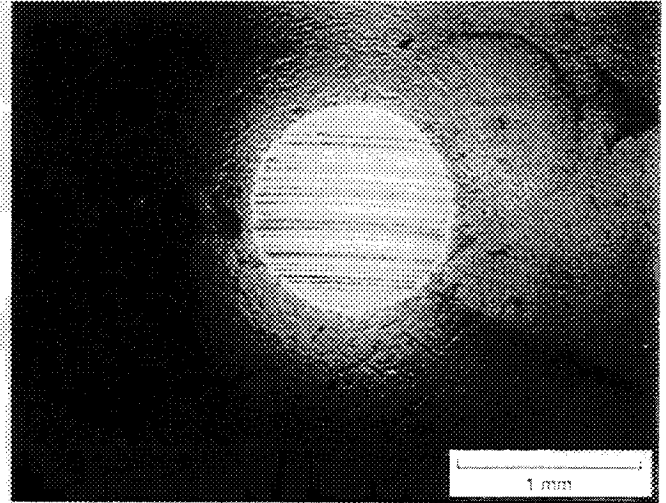


Figure 4.—Typical wear scar (600 N, 100 rpm, vacuum, PFPE Z-25, after 366 m of sliding).

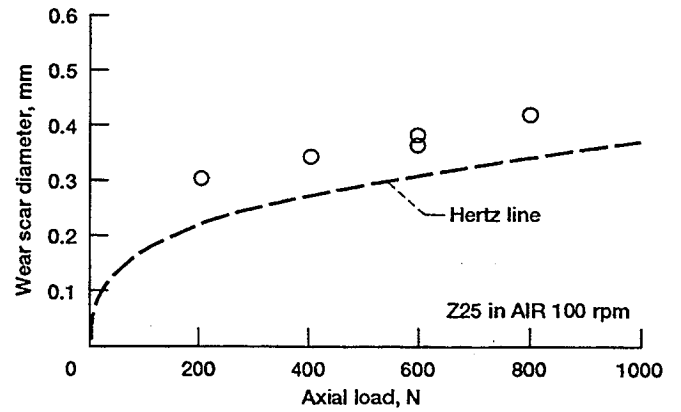


Figure 5.—Effect of axial load on wear scar diameter.

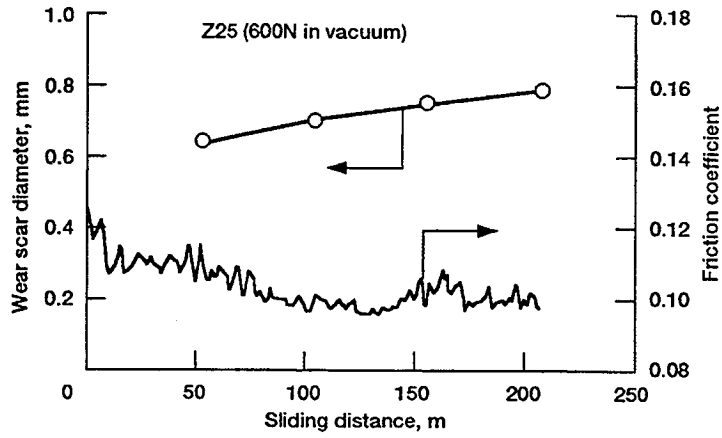


Figure 6.—Typical example of friction and wear results.

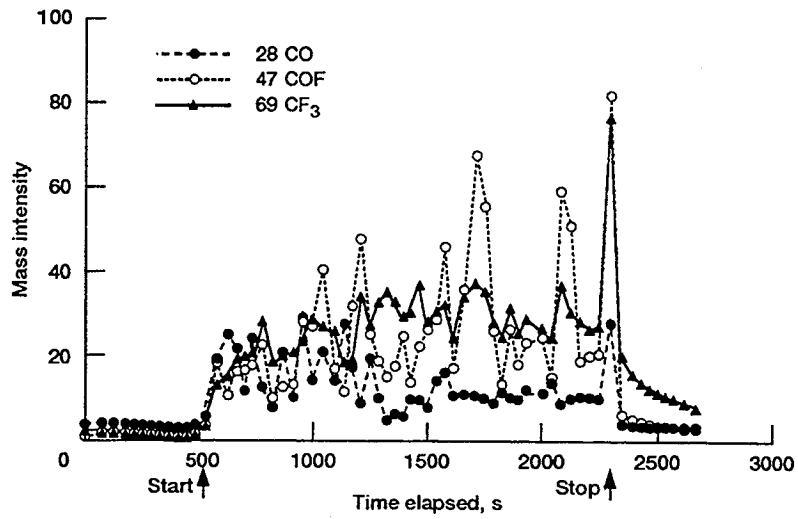


Figure 7.—Typical mass intensities as a function of test time (PFPE Z-25, vacuum, 200N, 100 rpm).

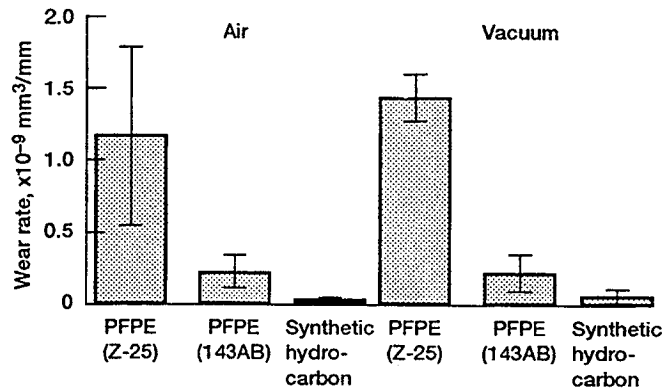


Figure 8.—Wear rates for three commercial aerospace lubricants in air and vacuum (25 °C, 200N load, 100 RPM)

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